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EXPERIMENTAL STUDY ON THE PERFORMANCE OF DUAL-EVAPORATOR REFRIGERATION SYSTEM WITH AN EJECTOR

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ABSTRACT

Experimental investigation on the performance of dual-evaporator refrigeration system with an ejector has been carried out. In this study, HFC-134a is chosen as a working fluid. Test setup is composed of compressor, condenser, expansion valves, two evaporators, and ejector which is installed between two evaporators. The condenser and two evaporators are made as concentric counter-flow type heat exchangers. Experiments are performed by changing the inlet and outlet temperatures of secondary fluids entering the condenser, high-pressure evaporator and low-pressure evaporator at constant compressor speed condition. When the external conditions are fixed, results show that coefficient of performance (COP) increases as the inlet temperature of the other evaporator becomes higher. It is also shown that the COP decreases as entrainment ratio of the ejector increases. For the same conditions, the COP of dual-evaporator refrigeration system with an ejector is superior to that of the single-evaporator vapor compression system by 3 to 6%.

NOMENCLATURE

b	Diffuser section outlet
COP	Coefficient of performance
C_p	Specific heat [kJ/kgK]
h	Enthalpy[kJ/kg]
i	Nozzle section outlet
j	Mixing section outlet
k	Constant area section outlet
m	Motive fluid
$\dot{m}$	mass flow rate [g/s]
P	Pressure [kPa]
$\dot{Q}$	Capacity [kW]
s	Suction fluid
T	Temperature [°C]
w	Entrainment ratio
$\dot{W}$	Work [kW]
x	Quality

Subscripts

b	Secondary fluid (Brine)
c	Condenser
comp	Compressor
e	Evaporator
e1	High-pressure evaporator
e2	Low- pressure evaporator
ej	Ejector
i	Inlet
o	Outlet
w	Secondary fluid (Water)

1. INTRODUCTION

A domestic refrigerator is used to hold a freezer temperature at about −18°C and food compartment temperature near 4°C, however, these two different levels of temperature are achieved by only one evaporator. A single-evaporator, located in freezer compartment, must cover both the food and freezer compartments. Therefore, it must operate at a lower pressure required by freezer section, which is much lower than that required by food compartment. In this case, due to a high pressure difference between evaporator and condenser, the compressor work is relatively large, which results in degraded performance.

The simplest way to reduce the compressor work and to achieve better performance is to make two independent refrigeration cycles with different evaporation temperatures, which is called 'dual-loop cycle'.[1] As the pressure rise of the refrigeration cycle is smaller, the compressor work decreases. Simulation results showed an improvement of 19% for a dual-loop cycle using R-22,[2] however, in spite of this advantage, the major problem with this system is its high cost. Because of the cost, interest has been given to another refrigeration cycle with two evaporators, 'single-loop cycle', which has only one compressor and two evaporators. Much attention is given to dual-evaporator refrigeration cycles in these days.[3-4] If there are two-independent evaporators and the temperature of each evaporator is controlled separately, several advantages such as high energy efficiency and low power consumption for defrosting can be taken. One additional advantage is a possibility to achieve a comfortable environment in a refrigerator, because the stink and the moisture are not spread out from the food compartment to freezer compartment.

In this stream of research, dual-evaporator refrigeration cycle with an ejector was mainly focused on in this study. This is a kind of single-loop cycle, which has two evaporators at different pressure levels and an ejector combining the outlet of two evaporators. The role of the ejector is a pre-compression. In the ejector cycle, the inlet pressure of the compressor increases and as a result the compressor work reduces as compared to the standard refrigeration cycle. In this paper, the performance of a dual-evaporator refrigeration system with an ejector is experimentally investigated using R-134a as a refrigerant.

2. DUAL-EVAPORATOR REFRIGERATION SYSTEM WITH AN EJECTOR

Two different types of dual-evaporator refrigeration systems with an ejector are introduced in this study. The first system is shown in Fig. 1 (a), which is called 'saturated vapor driven ejector cycle', In this study, subcooled liquid at the exit of condenser is expanded at the first expansion valve and 2 phase refrigerant mixture is flowing to the first evaporator (high-pressure evaporator). In the first evaporator, the refrigerant is partly evaporated and it is separated into vapor and liquid in the separator. The saturated vapor from the separator becomes a motive fluid for the ejector and the saturated liquid in the separator is expanded at the second expansion valve to the freezer pressure.

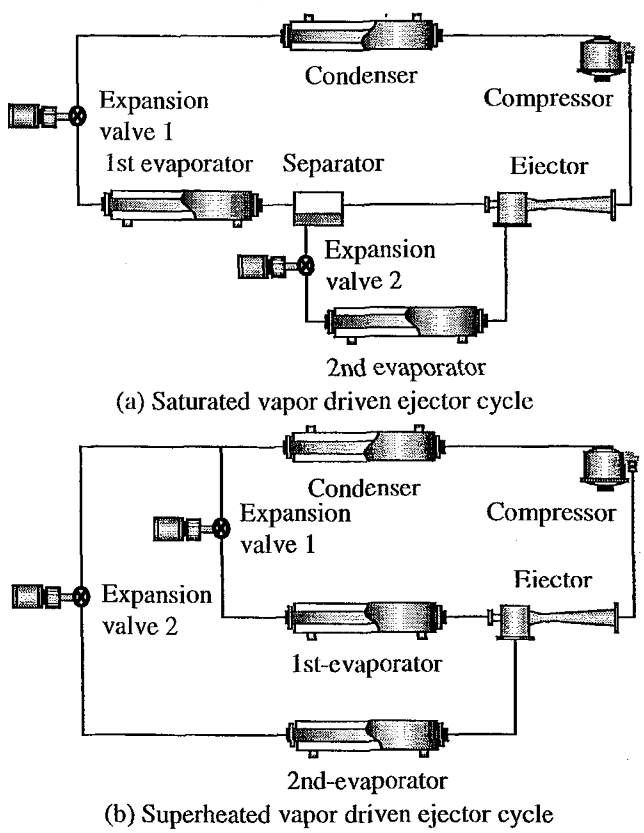


Fig. 1 Experimental setup for dual-evaporator refrigeration system with an ejector.

The vapor evaporated in the second evaporator (low-pressure evaporator) enters the ejector as a suction fluid. In the ejector, the motive flow and suction flow is well mixed in the mixing section of the ejector and this pre-compressed refrigerant is sent to the compressor.

The second system called 'superheated vapor driven ejector cycle' is shown in Fig. 1 (b), the cycle is almost the same with the saturated vapor driven ejector cycle except that the separator doesn't exist. After condensation, subcooled refrigerant is distributed into two streams; one is to the first evaporator (high-pressure evaporator) and the other is to the second evaporator (low-pressure evaporator). In the first evaporator, refrigerant is evaporated to a superheated vapor and this superheated vapor is used as a motive fluid in the ejector, while the refrigerant evaporated in the second evaporator becomes a suction fluid in the ejector. The refrigerant leaving the ejector is sent to the compressor.

3. EXPERIMENTAL APPARATUS AND TEST PROCEDURES

3.1 Ejector

An ejector is a device which entrains the low pressure suction flow by using the high pressure motive flow. Fig. 2 shows a basic structure of a standard ejector and the pressure change of the flow inside an ejector.[5] Motive fluid (m) of relatively high pressure enters the motive nozzle, through which it expands to produce a low pressure at the motive nozzle outlet (i). The high velocity motive stream entrains the suction fluid (s) of relatively low pressure into the mixing chamber. The velocity of mixed fluid (j) is generally supersonic. Within a constant area section normal shock wave is then generally produced, creating a compression effect, and the velocity of the mixed fluid is reduced to be subsonic. Further compression of the fluid is achieved as the combined streams flow through the diffuser section. Because the ejector has no moving or rotating part, it has many advantages such as simplicity, reliability and low cost. There have been many trials to put an ejector to a refrigeration cycle; Stoecker (1958) used the steam jet ejector as the expansion device in a refrigeration cycle[6] and Chen & Hsu (1987) used an ejector as a substitute for a compressor.[7] In this paper, the ejector is used in the dual-evaporator refrigeration system to reduce the compressor work by raising the pressure of the compressor suction gas.

3.2 Experimental Apparatus and Test Conditions

Fig. 3 shows experimental setup for dual-evaporator system with an ejector. The experimental setup is mainly composed of compressor, condenser, two expansion valves and two evaporators. On/off valves are used to switch from one to the other dual-evaporator refrigeration system and two needle valves are used to control the mass flow rate of the system. For the purpose to cool the condenser and to control the outlet condition of the condenser, secondary-fluid loop is used which is composed of pump, rotameter, reservoir and heat exchanger which is connected to a chiller. An electric heater is installed in the secondary-fluid loop to control the outlet condition of the evaporator and the mixture of ethylene glycol and water (40/60 by volume) is used as a secondary fluid.

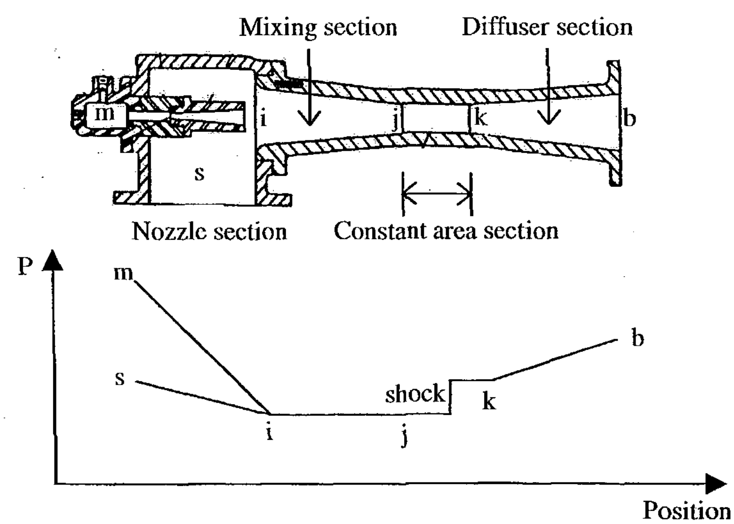


Fig. 2 Schematic view of the structure of an ejector and pressure change inside the ejector.[5]

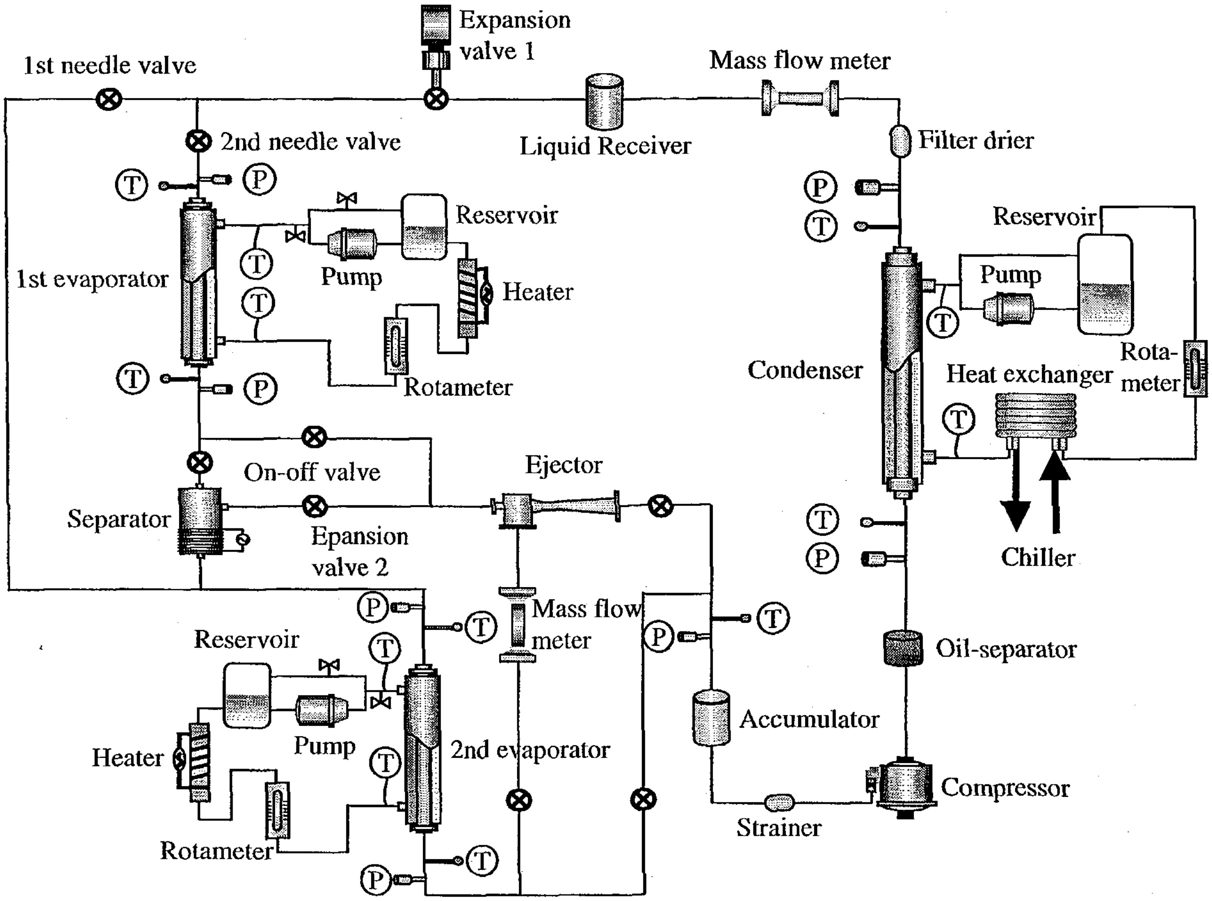


Fig. 3 Schematic diagram of experimental setup for dual-evaporator refrigeration system with an ejector.

The performance tests are carried out to obtain the variation of compressor work and COP with respect to external condition changes of the inlet temperatures of secondary fluids entering two evaporators. Table 1 shows secondary fluid temperatures for the performance test of dual-evaporator refrigeration system with an ejector. During the tests, superheating and subcooling were maintained to 8°C and the system performances are recorded when the variation of temperature, pressure and mass flow rate of the system is smaller than ±0.5°C, ±5 kPa and 0.2 g/s respectively.

4. RESULTS

4.1 Saturated Vapor Driven Ejector Cycle

4.1.1 Performance with respect to inlet temperature of secondary fluid

Fig. 4 (a) shows a refrigerant enthalpy difference between compressor inlet and outlet as a function of secondary fluid inlet temperature at low-pressure evaporator. In this test, the secondary fluid inlet temperatures of

Table 1 The secondary fluid temperatures of in this study.

$T_{w,c}$		$T_{b,e1}$		$T_{b,e2,i}$
Inlet	Outlet	Inlet	Outlet	
26°C	33°C	11°C	7°C	-4°C, 0°C, 4°C
		16°C	12°C	
31°C	38°C	11°C	7°C	-4°C, 0°C, 4°C
		16°C	12°C	

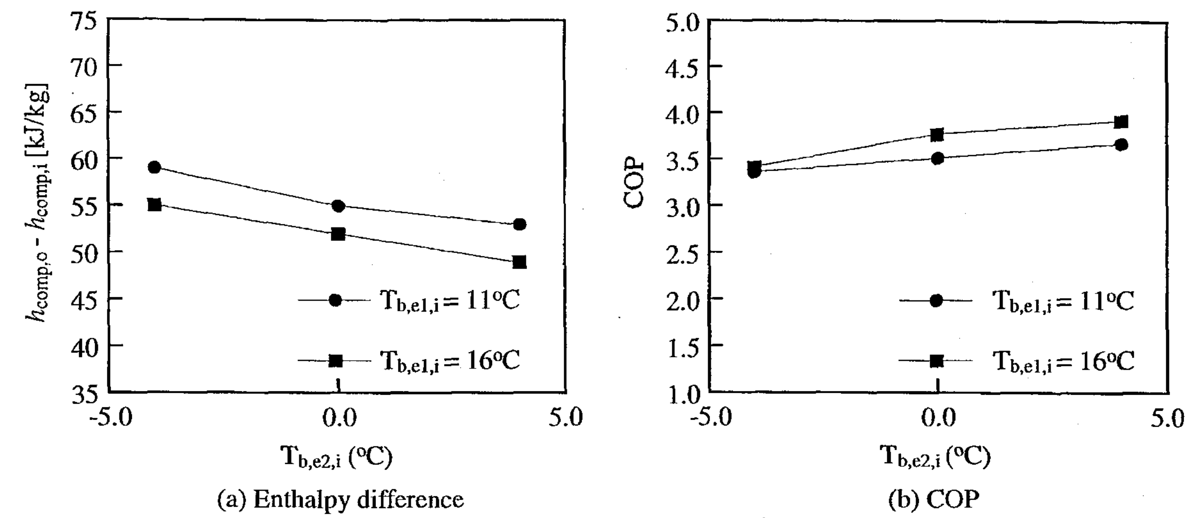


Fig. 4 Variations of enthalpy difference between compressor inlet and outlet and COP with respect to secondary fluid inlet temperature at low-pressure evaporator. ($T_{w,c,i}$ = 26°C, $\Delta T_{w,c}$ = 7°C, $T_{b,e1,i}$ = 11°C, 16°C, $\Delta T_{b,e1}$ = 4°C)

condenser and high-pressure evaporator are maintained constant and temperature difference between secondary fluid inlet and outlet is maintained as 7°C.

$$\dot{m}_{cj,m,i} h_{cj,m,i} + \dot{m}_{cj,s,i} h_{ej,s,i} = \dot{m}_{cj,o} h_{cj,o} \tag{1}$$

The enthalpy difference between compressor inlet and outlet decreases with an increase of secondary fluid inlet temperature of low-pressure evaporator or high-pressure evaporator when the other evaporator temperature is maintained constant. This phenomenon is resulted from the energy conservation of ejector inlets and outlet, which is shown in equation (1). If the mass flow rate and motive or suction flow enthalpy are maintained to be constant, the enthalpy of mixed flow at ejector outlet is in proportion to the enthalpy of the other flow (motive or suction flow). Since the compressor outlet condition is maintained constant and the enthalpy of the mixed flow is approximately the same as the enthalpy of the mixed flow at ejector outlet, secondary fluid inlet temperature of low-pressure or high-pressure enthalpy is inversely proportional to the enthalpy difference between compressor inlet and outlet.

This influences to the relation COP and secondary fluid inlet temperature of high-pressure or low-pressure evaporator. Fig. 4 (b) shows the variation of COP. The cooling capacity of high-pressure and low-pressure evaporator and compressor work is shown in equation (2) and (3).

$$\dot{W}_c = \dot{m}_{comp} (h_{comp,o} - h_{comp,i}) \tag{2}$$

$$\dot{Q}_{el} = \dot{m}_{c1} \int_{T_{c1,i}}^{T_{c1,o}} C_{p,e1} dT, \qquad \dot{Q}_{el} = \dot{m}_{c2} \int_{T_{c2,i}}^{T_{c2,o}} C_{p,e2} dT \tag{3}$$

Since the enthalpy difference between compressor inlet and outlet represents the compressor work per unit mass of refrigerant, the enthalpy of the mixed flow at ejector outlet and COP defined as in equation (4) increases when the secondary fluid inlet temperature of high-pressure or low-pressure evaporator gets high,.

$$\text{COP} = \frac{\dot{Q}_{el} + \dot{Q}_{e2}}{\dot{W}_c} \tag{4}$$

4.1.2 Performance with respect to high-pressure evaporator outlet quality

Fig. 5 shows a variation of COP with respect to the quality of refrigerant coming from high-pressure evaporator. As the high-pressure evaporator outlet quality increases, COP also increases. The meaning of the quality increase at the high-pressure evaporator outlet is that the mass flow rate of vapor phase which is related with the motive flow of ejector increases and that of liquid phase which is connected with the suction flow of ejector decreases. Equation (5) shows energy balance derived from equation (1) at slightly perturbed condition by the quality increase of high-pressure evaporator outlet.

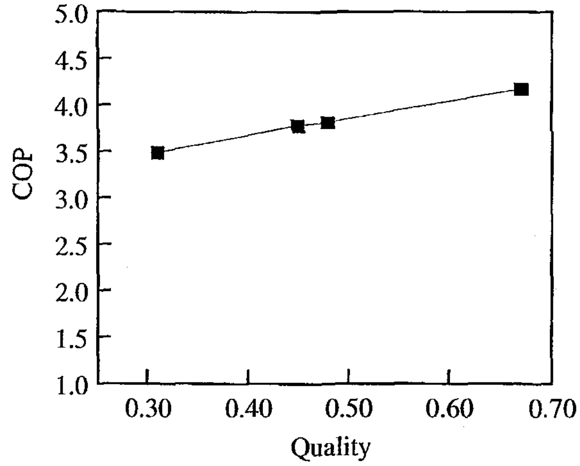


Fig. 5 Variation of COP with respect to the quality at high-pressure evaporator outlet. ($T_{w,c,i} = 26°C$, $\Delta T_{w,c} = 7°C$, $T_{b,e1,i} = 11°C$, $\Delta T_{b,e1} = 6°C$, $T_{b,e2,i} = 0°C$, $\Delta T_{b,e2} = 4°C$)

$$(\dot{m}_{ej,m,i} + \Delta\dot{m}_{ej})h_{ej,m,i} + (\dot{m}_{ej,s,i} - \Delta\dot{m}_{ej})h_{ej,s,i} = \dot{m}_{ej,o}(h_{ej,o} + \Delta h_{ej,o}) \tag{5}$$

By subtracting equation (1) from equation (5) equation (6) is obtained, In equation (6), since $h_{ej,m,i} - h_{ej,s,i}$ is always positive, $\Delta h_{ej,o}$ should be always positive. This means that the enthalpy of ejector outlet and that of compressor inlet increases and compressor work decreases. It is well known that the outlet pressure of ejector is a function of the ratio of the mass flow rate of suction flow to the mass flow rate of motive flow.[4]

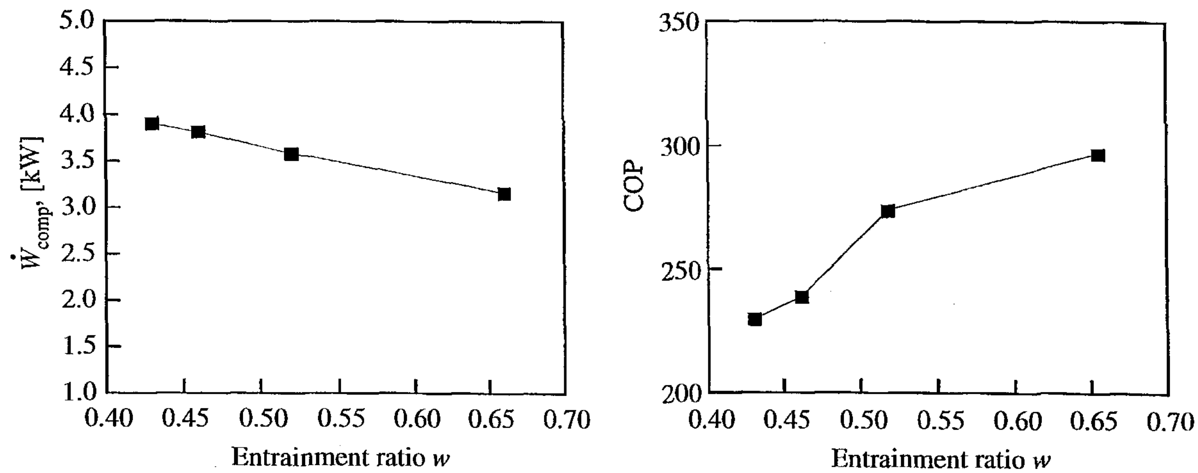
$$\Delta\dot{m}_{ej}(h_{ej,m,i} - h_{ej,s,i}) = \dot{m}_{ej,o}\Delta h_{ej} \tag{6}$$

Fig. 4 shows an increase of COP as the quality at high-pressure evaporator outlet increases. However, if the quality at high-pressure evaporator outlet is too high, cooling capacity of low-pressure evaporator reduces to zero. Thus, cooling capacity can be insufficient to the load of the freezer section because of the decreased mass flow rate.

4.2 Superheated Vapor Driven Ejector Cycle

4.2.1 Performance with respect to entrainment ratio

Entrainment ratio, w, is defined as a ratio of the mass flow rate of suction flow to the mass flow rate of motive flow as shown in equation (7)



(a) Compressor work (b) COP

Fig. 6 Variations of compressor work and COP with respect to entrainment ratio. ($T_{w,c,i} = 31°C$, $\Delta T_{w,c} = 7°C$, $T_{b,e1,i} = 16°C$, $\Delta T_{b,e1} = 6°C$, $T_{b,e2,i} = 0°C$, $\Delta T_{b,e2} = 4°C$)

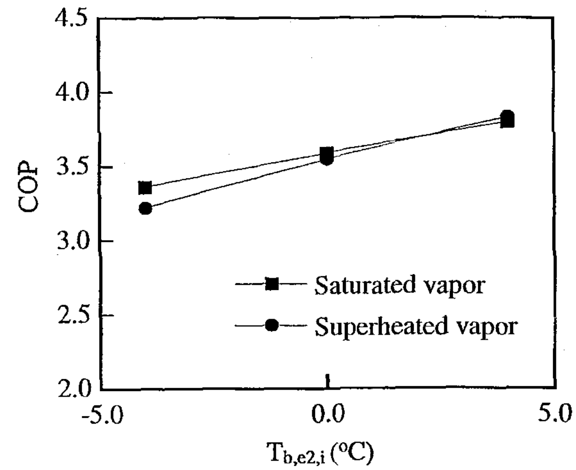


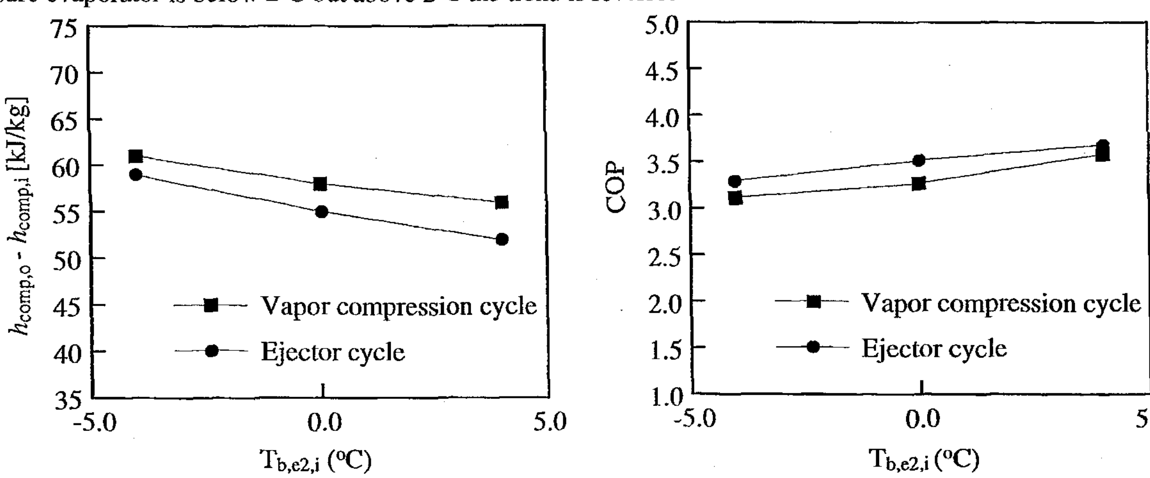
Fig. 7 Comparison of the COP of saturated vapor driven and superheated vapor driven ejector cycle with respect to the secondary fluid inlet temperature of low-pressure evaporator. ($T_{w,c,i} = 31°C$, $\Delta T_{w,c} = 7°C$, $T_{b,e1,i} = 16°C$, $\Delta T_{b,e1} = 6°C$)

$$w = \frac{\dot{m}_{ej,s,i}}{\dot{m}_{ej,m,i}} \tag{7}$$

The effect of the entrainment ratio on compressor work and COP is shown in Fig. 6. In the superheated vapor driven ejector cycle, refrigerant flow is divided into two streams after condensation. One stream is sent to the low-pressure evaporator and the other stream is sent to the high-pressure evaporator. As the mass flow rate of refrigerant through the low-pressure evaporator increases, entrainment ratio decreases and the ejector outlet enthalpy decreases as shown in equation (6). As a result, the compressor work increases and COP decreases. Fig. 6 (a) shows an increase of compressor work and Fig. 6 (b) represents a decrease of COP when the entrainment ratio increases.

4.3 Comparison of Saturated Vapor Driven Ejector Cycle and Superheated Vapor Driven Ejector Cycle

Fig. 7 compares the COP of saturated vapor driven ejector cycle and superheated vapor driven ejector cycle with respect to the secondary fluid temperature entering low-pressure evaporator. In saturated vapor driven ejector cycle, the total mass flow rate is 5.20 g/s, and the quality at the outlet of high-pressure evaporator is 0.4, in superheated vapor driven ejector cycle, the total mass flow rate is 5.65 g/s and the entrainment ratio is 0.5. COP of the saturated vapor driven ejector cycle is slightly higher when the secondary fluid temperature entering low-pressure evaporator is below 2°C but above 2°C the trend is reversed.



(a) Enthalpy difference (b) COP

Fig. 8 Variations of enthalpy difference between compressor inlet and outlet and COP change with respect to secondary fluid temperature of entering low-pressure evaporator. ($T_{w,c,i} = 31°C$, $\Delta T_{w,c} = 7°C$, $T_{b,e1,i} = 16°C$, $\Delta T_{b,e1} = 6°C$, $T_{b,e2,i} = 0°C$, $\Delta T_{b,e2} = 4°C$)

4.4 Comparison of Dual-Evaporator Refrigeration Ejector Cycle and Single-Evaporator Refrigeration Cycle

In Fig. 8, the enthalpy difference between compressor inlet and outlet and the COP of single-evaporator vapor compression cycle is compared to those of dual-evaporator refrigeration cycle with an ejector (saturated vapor driven ejector cycle). For these two cycles, the comparison was made when the secondary fluid temperatures at the condenser inlet and at low-pressure evaporator inlet are maintained the same. The secondary fluid inlet temperature of high-pressure evaporator is also maintained to a constant value.

In dual-evaporator refrigeration cycle with an ejector (saturated vapor driven ejector cycle) and single-evaporator vapor compression cycle, evaporator outlet pressure (low-pressure evaporator in case of ejector cycle) is the same, but the compressor inlet pressure of ejector cycle is higher than that of vapor compression cycle due to the pre-compression by the ejector. As a result, in Fig. 8 (a), the enthalpy difference between compressor inlet and outlet of ejector cycle is 5% lower than that of vapor compression cycle. As shown in Fig. 8 (b), the COP of ejector cycle is superior to that of vapor compression cycle by 3 to 6%.

5. CONCLUSIONS

Experimental investigation on the performance of dual-evaporator refrigeration ejector cycle has been carried out and the conclusions of this study can be summarized as follows

(1) In the saturated vapor driven ejector cycle, as the secondary fluid inlet temperature of low-pressure evaporator or high-pressure evaporator becomes high, the compressor work decreases and the COP increases.
(2) In the saturated vapor driven ejector cycle, the compressor work decreases and the COP increases as the quality of high-pressure evaporator outlet increases.
(3) In the superheated vapor driven ejector cycle, the compressor work increases and COP decreases as the entrainment ratio increases.
(4) COP of the saturated vapor driven ejector cycle is slightly higher than that of the superheated vapor driven ejector cycle when the secondary fluid inlet temperature of low-pressure evaporator is low.
(5) Compressor work of dual-evaporator refrigeration system with an ejector decreases by 5% compared to that of single-evaporator vapor compression cycle and the COP is higher 3 to 6%.

ACKNOWLEDGMENTS

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